



Behaviour of big bearings: Polymer coated bearings in comparison to Babbitt bearings

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Introduction

This paper is motivated by problems occurring with large slide bearings. To optimize these bearings, a generally applicable simulation method for hydrodynamic bearings has been developed. A comparison between computer simulations and measurements has shown a very high concordance.

The combination of findings from both measurements and simulations provides an exact understanding of the processes in the bearing. The use of polymeric coatings, whose behavior slightly differs in hydrodynamic pressure generation from that of white metal, results in new, extended possibilities for modern bearing design.

1 Technology of Large Bearings

The rotors in large machines such as generators or motors with some ten to several hundred MW power typically weight between 50 and 3,000 tonnes. This weight is borne by conventional, hydrodynamic slide bearings. Machines of several hundred MW are generally designed vertically and have only one axial bearing, which bears the complete load.

Depending on the rotational speed, the thrust bearings for machines of this size may reach diameters of 2 meters or more. The bearing pads have side lengths of half a meter to one meter. The thrust bearing of the largest water power plant the Three Gorges Project with an installed power of 18200 MW has the spectacular external diameter of 5.2m, and with its 24 bearing pads, which has been tested to a load up to 6000 tonnes. Fig. 1 shows test of the bearing pads.



Fig. 1: Large thrust bearing in a test stand with an axial load up to approx. 3000 tonnes.

These large dimensions result in an interesting problem concerning the thermal behavior of the thrust bearing pads. As the bearings are flooded with machine oil, the pads are embedded in relatively cool oil whereas the surface of the oil film may become rather warm. Unless counteractive measures are initiated, thermal crowning of more than $100\mu\text{m}$ in height may result. Although such a deformation seems to be rather negligible in relation to the pad lengths of 850mm , it suffices to reduce the hydro-dynamically active surface so much that the bearing's load bearing capacity is exceeded with damaging consequences. To classify the significance of thermal deformation, it is important to know that the minimum oil film thickness of these bearings has a range of 30 to $50\mu\text{m}$. In addition, machine oil only develops noteworthy pressure values at an oil film thickness between 20 and $200\mu\text{m}$. This physical fact applies to all bearing sizes, even for thrust bearings with diameters of 55mm as they are used e.g. in a diesel engine or smaller. In this case, crowning would be reduced to 2 to $3\mu\text{m}$ on the same conditions, which is not critical. In general, a crowning height of at maximum half of the minimal oil film gap is recommended.

A possibility of counter balancing the crowning problem of large bearing pads is the so-called "Double Layer" pad technology developed by the former ABB, today ALSTOM. The bearing is made up from a rigid load carrying layer with a constant temperature and a flexible layer placed on top of the first one, supported on elastic pins. For further details, please refer to [4]. This solution has been proven and tested for 22 years now; however, it requires highly precise manufacture.

More than 50 years ago, some bearing manufacturers met with problems in producing thrust bearing pads with an evenness of approximately $10\mu\text{m}$. To avert this problem, Russian manufacturers have pressed a Teflon layer onto a basic steel body. Teflon, a ductile and heat-resistant material, is grinded to the final evenness during operation by the runner, which is the bearing pads' counter piece. Apart from that, the material properties of Teflon improve the bearings' efficiency insofar that the glass transition temperature is approximately twice as high as with the conventional white metal coating. Teflon also forms a thermal isolating layer between the hot oil and the cool steel pad, making the temperature gradient in the basic steel body quite small. Last but not least, Teflon is more tolerant in contact behavior with the runner, in case that the oil film will be interrupted. This may be the case when switching off machines with a free run out. When the speed falls below a critical value, the oil film breaks down. Due to the low friction coefficient between Teflon and steel with mixed friction, less heat is generated than with white metal. Another aspect is the significantly smaller E-modulus of Teflon, which may have a decisive impact on the oil film generation; an aspect that will be discussed below.

2 Experiment

A test stand was set up to compare thrust bearing pads with more than 30 different polymer coatings with each other as well as with white metal. This test stand consists of a rotor driven by an electric motor. A retainer containing two hydraulic cylinders holds two thrust bearing pads that are axially symmetrical. Due to the pressure in the hydraulic cylinders the pads can be charged with an axial load. The pressure cylinder has a spherical support so that the bearing body tilts in a way that the balancing of moments is reached naturally. To exactly measure the pressure progression in the oil film, a high-resolution pressure measuring system is built into the rotor.

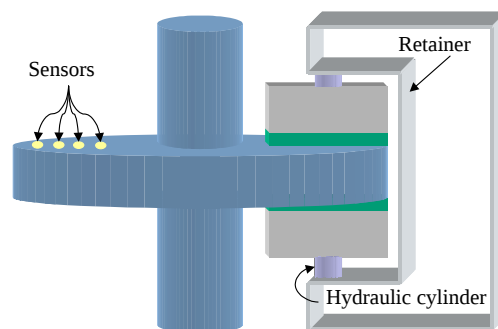


Fig. 2: Bearing test stand

The rotor disk is 720mm in diameter and reaches a speed up to 1500 r.p.m. The bearing pads have a surface of approx. 100 x 100mm. The setup of the test stand is flexible, enabling quick change of pad with different configurations as well as the equipment of measuring devices.

Four pressure sensors mounted in the runner have a resolution of 2,048 measuring points per revolution. Thus, 114 measuring points are available above the pad per channel and revolution. The signals may be interpolated three-dimensional. The examination of the measured values and the comparison with the simulation is handled in the chapter after the next one.

3 Simulation

To understand the behavior of different thrust bearings, it is recommendable to conduct a large number of measurements. However, the correlations are easier to understand with parallel simulations. On the other hand, it is always necessary to verify the simulations so that predictions and conclusions can be ensured.

The calculation is completely based on the Finite Element method. The pressure built up in the oil film is described by the Reynolds equation (1). The derivation is described in [6], so this section provides only a short introduction. On the right hand side of the equation (1) the origin of the hydrodynamic pressure buildup is described. In the observed case, this is only the wedge action. A fluid particle arrives in the converging gap between the stationary pad and the rotating disk. In addition to the relative motion $r \cdot \omega$ a converging gap $\partial h / \partial \theta$, i.e. a wedge, is necessary to build up a pressure. The gap-function is described by θ .

$$\underbrace{\frac{\partial}{\partial r} \left(\frac{r \cdot h^3}{\mu} \frac{\partial p}{\partial r} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial \theta} \right)}_{\text{pressure distribution}} = \underbrace{6r\omega \frac{\partial h}{\partial \theta}}_{\text{wedge action}} \quad (1)$$

The left hand side part of the equation describes the pressure distribution. This is essentially determined by the pressure gradients in tangential and radial direction, $\partial p / \partial \theta$ and $\partial p / \partial r$. The gap-function h has an essential impact, too. With high loads, a variable dynamic viscosity must also be taken into account for an exact calculation, as in that case the friction in the fluid is so high that the local warming modifies the viscosity. As a rule of thumb, we can assume a bisection of the viscosity per 20K. To accommodate this temperature change, the energy equation (2) is applied, which is equally described in [5] and [6]. The principal setup is the same as with the Reynolds equation. The driving parameters of temperature change by viscous dissipation are the relative movement and the pressure gradients $\partial p / \partial \theta$ and $\partial p / \partial r$. The temperature distribution, expressed by the convective heat, i.e. the left hand side of the energy equation (2), is affected by the oil density ρ , the heat capacity c_p , the angular speed ω and the gap-function h .

$$\frac{1}{2} \cdot \omega \cdot \rho \cdot c_p \cdot h \cdot \left\{ \left[1 - \frac{h^2}{6 \cdot \mu \cdot \omega \cdot r^2} \frac{\partial p}{\partial \theta} \right] \frac{\partial T}{\partial \theta} - \frac{h^2}{6 \cdot \mu \cdot \omega} \left(\frac{\partial p}{\partial r} \right) \frac{\partial T}{\partial r} \right\} = \frac{\mu \cdot \omega^2 \cdot r^2}{h} + \frac{h^3}{12 \cdot \mu} \left[\frac{1}{r^2} \left(\frac{\partial p}{\partial \theta} \right)^2 + \left(\frac{\partial p}{\partial r} \right)^2 \right] \quad (2)$$

The simultaneous solving of both equations, see [3], completely covers the processes in the oil film; only the adjacent surfaces need to be observed closer.

The pressure and temperature buildup in the oil film have an impact on the neighboring bodies, the runner and the bearing pad. The modeling, especially that of the pads, must exactly reproduce the actual setup as the oil film is extremely sensitive to changes in the lubricating gap. The oil film properties are determined by an ALSTOM in-house software development and the structural properties of the neighboring bodies is solved by the commercial FEM program ANSYS. Fig. 3 shows the geometry model for the input in ANSYS.

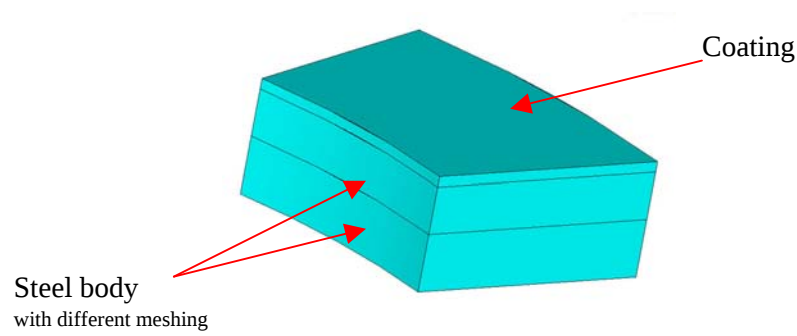


Fig. 3: FE model of the thrust bearing pad

The entire simulation procedure is shown in as a diagram.

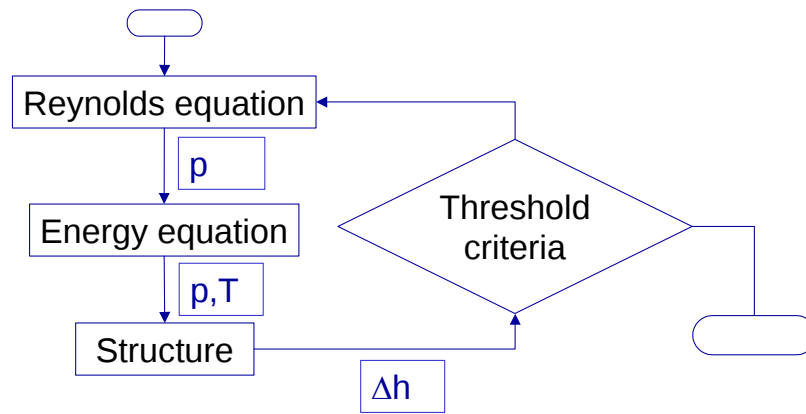


Fig. 4: Flow chart of the simulation procedure

The factors influencing the oil film, the pressure buildup and thus the properties of the bearings will be discussed in the following sections.

4 Comparison between Simulation and Measurement

The simulation method described is rather complex and also very sensitive to inaccurate input. The exact verification of this method is thus very important. For this paper, two representative measurements were chosen: one with white metal coating and one with, ALSTOM's polymer coating PolyPad®. The last measurement explicitly reveals the autonomous behavior of the polymer in comparison to the white metal coating.

At first the white metal pad is observed. For a direct comparison between test and measurement the pressure sensor signals are compared directly, in order to emphasize the differences.

The pressure progression is acquired at the radii 287, 305, 323 and 341 over the bearing pad. The continuous lines in Fig. 4 stand for the measured values; the dashed lines stand for the simulated values. For the same radius, the same symbols are used for both the measurement and the calculation, i.e. the symbol * for the radius of 287mm. The highest pressure is built up in the center of the bearing, acquired by the sensors on the radii 305mm and 323mm. The simulation comes very close to the pressure progression and the maximum values of the four sensors; however, the increase of the simulated pressure, i.e. the dashed line, is almost linear at the beginning (see Fig. 4), while the actually measured pressure, i.e. the continuous line, rather tends to "sag" slightly. In general, we may say that the simulation reproduces the actual pressure progression very exactly, and the presented method completely comes up to expectations.

The selected measurement is taken at 964 r.p.m. that equals to an average speed of 31.8m/s. The load is 31.6kN, which corresponds to a mean load of 2.95 MPa.

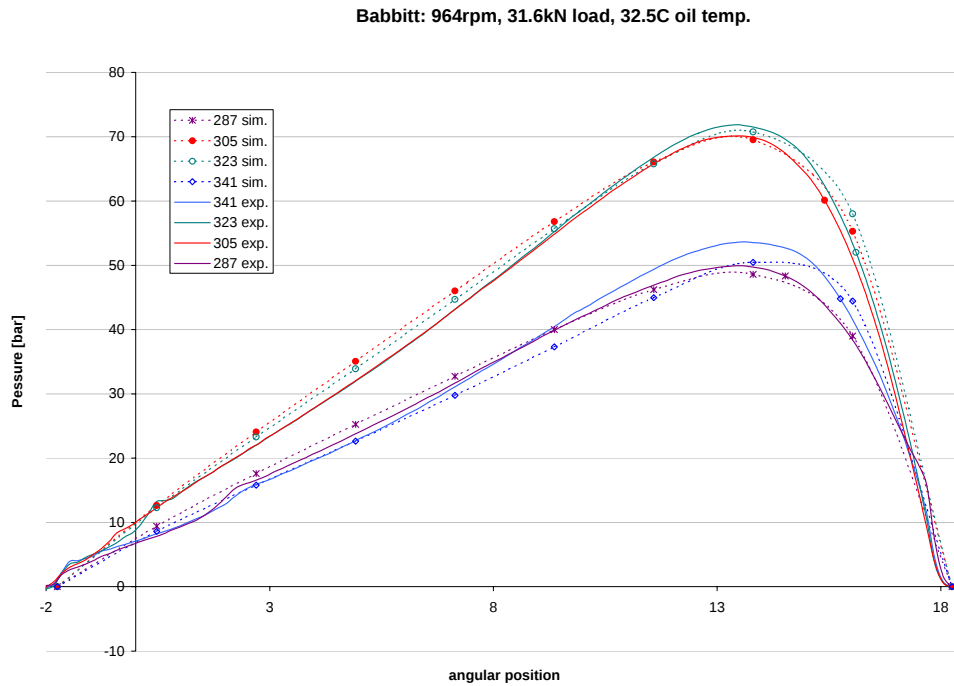


Fig. 4: Comparison between simulation and experimental data for white metal charged with a load of 32.6kN.

The next task is to examine and compare the behavior of a thrust bearing pad coated with PolyPad® specially optimized for slide bearing applications instead of white metal. Relating to the pressure buildup, there is a decisive difference between synthetic materials and white metal.

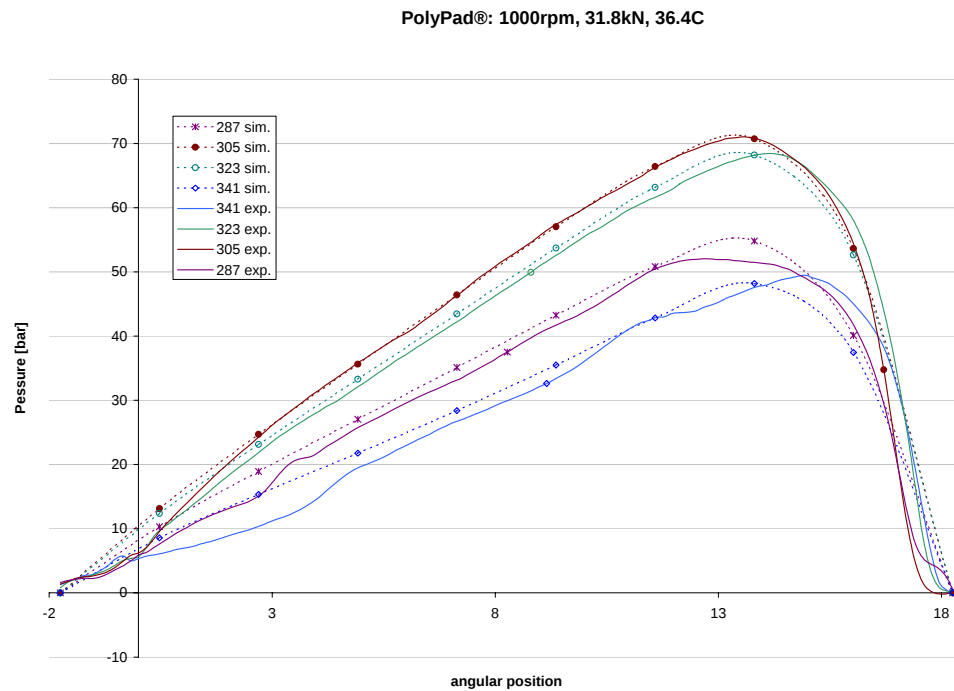


Fig. 5: Comparison between simulation and experimental data for polymer coating charged with a load of 31.8kN.

The Youngs-modulus of such a bearing coating, whether it is based on Teflon or Peek like PolyPad®, remains below that of white metal by one to two orders of magnitude, which has a decisive influence on the pressure progression.

The simulation of the PolyPad® bearing also very closely follows the measured values, taking into account the changed pressure progression. The increase is no longer linear but ball-shaped. The maximum values with 71.0bar are below those of white metal with 71.9bar. The maximum values for PolyPad® at 13.54° have shifted slightly as well compared to the white metal peak at 13.47°. Compared to a previous publication [2] the polymer coating could be improved in a way to perform in a similar way a white metal remaining significant advantages like:

- Heat resistant to over 200°C for continuous operation.
- No significant wear or creep.
- Very good geometrical integrity, better than Teflon®.
- No peel off from the base body due to full penetration of the PolyPad® material in the porous metal grid.
- Very low friction at startup, no high pressure jacking required.
- High pressure jacking is not required but applicable.
- No maximum resting time. The machine can start without jacking after long resting periods.
- PolyPad® is designed for high number of starts, at least 1500 p.a.
- Usually there is no maintenance.
- Projected lifetime depends on the load and number of starts, usually over 20 years if the oil remains clean.
- Double load capacity compared to common white metal pads.
- Optimized bearing geometry can save more than 30% of bearing losses compared to white metal.
- High operational safety, especially for operation in temporary mixed lubrication.
- Can be a one by one replacement for existing taper-land, tilting pad or spring-bed bearing pads.
- The price of a PolyPad® is higher than a standard white metal pad. Nevertheless, the system price is often cheaper than for a standard bearing.

The simulation technique for hydrodynamic bearings were verified with many datasets of the test stand and for a generator bearing with 2.2m in diameter. Several bearing pads with white metal, PFA and PTFE coating and bronze were studied. This elaborated know-how makes it possible to predict bearing behavior through simulation and therefore optimize the bearing for each individual application according to the customers requirements.

A PolyPad® bearing is applied in the power plant Mühleberg [7] since May 2003 as replacement for an existing bearing which was damaged during off design operation. The bearing is running well. It can even be operated at higher cooling water temperatures during summer were the original bearing had to be switched off. Thereby, the machine productivity could be improved.



Fig. 6: PolyPad® application at Mühleberg: First check of the Bearing after 10 month of operation. The bearing does not show any deformation, damages or unusual wear marks.

5 Concluding Discussion of the Influencing Variables

The paper has shown that hydrodynamic bearing behavior can be simulated quite precisely, whether it is a conventional or a polymer coated bearing.

The above examination shows the impact of bearing coating material may have on the pressure buildup. The most important parameters are the coating properties. Of course, the bonding layer between polymer and steel body must not be neglected.

Another important factor that has not been taken into account in this paper is the type of pad support and the exact position of the pivot point, which determines the shape and size of the pressure progression in moving direction and the relation of the lateral pressure distribution.

The variation of the viscosity, which directly depends on the temperature, is another interesting parameter that cannot be discussed in the scope of this paper. It does not have a major impact on the pressure progression of the observed thrust bearing, although it has been taken into account so that the presented accuracy can be achieved. The viscosity especially affects the gap height and the power dissipation of the bearing.

Furthermore, the local oil film pressure is decisively affected by manufacture-dependent geometrical deviations; however, the load bearing capacity is hardly affected within certain limits. Axial tilting pad thrust bearings are rather insensitive in comparison to fixed pad or radial bearings, as the free adjustment of the lubricating gap and the tilting may have a balancing effect. Nevertheless, the pressure progression reveals surface unevenness of several micrometers, air pockets in the synthetic material or unevenness in the bonding layer between steel body and synthetic material – see also [1].

In conclusion it may be asserted that it is absolutely possible to exactly simulate the processes in a hydrodynamic bearing by means of state-of-the-art tools. The understanding of the bearing behavior reveals

new possibilities in bearing design and coating selection. The elaborated know-how made it possible to select a superior material for bearing applications.

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